PATENT SPECIFICATION

732,319

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COMPLETE SPECIFICATION

Improvements in Variable Speed Driving Mechanisms

We, WALLACE & TIERNAN, INCORPORATED, a Corporation organised and existing under the Laws of the State of Delaware, United States of America, of 100, West Tenth Street, Wilmington, Delaware, United States of America, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The present invention relates to variable speed driving mechanisms for driving a load at an average speed which is continuously variable over a wide range.

Many variable ratio driving mechanisms have been constructed in which the speed of an output shaft is variable over a wide range while the speed of an input shaft remains constant. In such previous mechanisms, the output shaft rotates continuously at all speed settings. The problem of providing a continuously variable output shaft speed with a fixed input shaft speed is mechanically very difficult, and the previous devices have been either very complex or they have included mechanical elements which required close dimensional tolerances. In either case, the construction and maintenance costs have been excessive.

Some loads, requiring operation at variable speeds, also require that the shaft driving them be continuously rotated. Many such loads which are at present driven with continuously rotating shafts may, however, be successfully operated with an intermittent drive, providing the average speed of rotation of the shaft is continuously variable over a wide range. Among the loads which may be successfully operated with an intermittent drive are feeding mechanisms for many chemical and industrial processes. For example, a device for feeding dry material, such as shown in the co-pending Application No. 11,731/53 (Serial No. 724,475), may be successfully operated by an intermittent drive for most of its applications.

According to the present invention, a variable speed driving mechanism includes a clutch interposed between a driving shaft and a coaxial driven shaft and engaged and disengaged by relative axial movements of the two shafts, and one of the shafts is reciprocated so as to engage the clutch for a proportion only of each revolution by means of a cam and follower device adjustable so as to vary the proportion of a revolution during which the clutch is in engagement. Thus the driven shaft is driven for only that portion of a revolution during which the clutch is in engagement and its average speed is reduced accordingly. In other words, the average speed of the driven shaft will be approximately the same proportion of the speed of the driving shaft as the proportion of each revolution during which the clutch is engaged.

By suitably adjusting the cam and follower device, the speed of the driven shaft may be correspondingly adjusted, and preferably the cam and follower device is so designed as to enable this adjustment to be carried out continuously over a range lying between 0 and 100 per cent of a revolution. Thus in one extreme position the clutch is not engaged at all and the driven shaft does not turn, while in the other extreme position the clutch is continuously engaged and the driven shaft turns at the same speed as the driving shaft. For intermediate positions of adjustment, the speed of the driven shaft is a corresponding fraction of that of the driving shaft.

Mechanism in accordance with the invention will now be described in more detail by way of example with reference to the accompanying drawings, in which:

Fig. 1 is a sectional view of the mechanism;

Fig. 2 is an elevational view of the internal parts of the mechanism of Fig. 1, showing

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the parts in a different operating position;

Fig. 3 is a developed view of the cam surfaces and the followers, showing diagrammatically the positional relationship of the follower rollers to the cam surfaces under various operating conditions;

Fig. 4 is an oblique view of the cam and follower mechanism; and

Fig. 5 is a sectional view similar to Fig. 1 illustrating a modified form of variable driving mechanism.

Referring to Figs. 1 to 4 of the drawings, there is shown a casing, which may be filled with oil. Journalled in the casing 1 is an input shaft 2 on which is fixed a gear 3. The gear 3 meshes with a gear 4 fixed on a shaft 5, hereinafter termed the driving shaft. A clutch plate 6 is mounted on the left-hand end of shaft 5, as viewed in the drawings. The clutch plate 6 co-operates with a driven clutch plate 7 mounted on a driven shaft 8, which is hollow and concentric with the shaft 5, and rotatably and slidably mounted on the shaft 5 and journalled in the casing 1. The driven shaft 8 has fixed thereon a gear 9 which meshes with a gear 10 fixed on an output shaft 11 journalled in the casing 1.

The end of driven shaft 8 opposite the clutch plate 7 carries a bearing 12. A spring 13 is retained in compression between the bearing 12 and the gear 4 and biases the driving and driven shafts for relative movement in a sense to hold the clutch plates 6 and 7 in engagement.

The clutch members 6 and 7 are intermittently engaged and disengaged by reciprocation of the driving shaft 5. This reciprocation of shaft 5 is produced by mechanism for producing reciprocation from continuous rotation, and which appears at the right-hand side of Figs. 1 and 2.

The right-hand end of shaft 5 is journalled in a wall of the casing 1. Within the casing 1 and spaced from the right-hand end, the shaft 5 carries a follower mechanism including a yoke 14 mounted on a pivot pin 15 which extends perpendicularly to the axis of shaft 5. The yoke 14 carries diametrically opposite, outwardly projecting pins 16, perpendicular to the pivot pin 15. Follower rollers 17 and 18 are journalled on the pins 16.

The rollers 17 and 18 co-operate as followers with a pair of concentric adjacent cylindrical cams 19 and 20. The cams 19 and 20 have complementary projections 19*a* and 20*a* and recesses 19*b* and 20*b*. The inner cam 20 is fixed to the casing 1. The outer cam 19 is angularly adjustable with respect to the cam 20. For the purpose of adjustment, the cam 19 is provided on its outer periphery with a worm gear 21 which co-operates with a worm 22 fixed on a shaft 23. Shaft 23 extends outside the casing 1. A bevel gear 24 and a hand wheel 25 are fixed on the shaft 23 outside the casing 1. The bevel gear 24 co-operates with another bevel gear 26 to drive a register mechanism 27. The roller followers 17 and 18 are wider than the cams 19 and 20, and the followers overlap both cams. Consequently, if a portion of a projection on one cam is radially adjacent a portion of a recess on the other, the projection, rather than the recess, determines the axial positions of the followers as they become radially aligned with those adjacent cam portions.

The clutch plate 7 has a larger diameter than the clutch plate 6. A stop plate 28 is mounted on one end of the casing 1 by means of studs 29. The stop plate 28 has an aperture 28*a* which is larger in diameter than the clutch plate 7.

When the parts are in the positions shown in Fig. 1, the clutch plates 6 and 7 are held in engagement by the spring 13. The clutch plate 7 is spaced from the stop plate 28. In the particular angular position of the rollers 17 with respect to cams 19 and 20 which is shown in Fig. 1, the assembly including the shafts 5 and 8, yoke 16 and rollers 17 is free to float between a position in which both the rollers 17 and 18 engage the cams 19 and 20, and a position in which the clutch plate 7 engages the stop plate 28. As long as that assembly floats freely between those two positions, the input shaft 2 remains in driving connection with the output shaft 8 through the clutch 6, 7. If the assembly is floating in the position where the clutch plate 7 engages stop plate 28, there will be no tendency for the clutch plates to separate, since the spring 13 is still effective to hold them together. Furthermore, there will then be no substantial friction between the clutch plate 7 and the stop plate 28 since there is substantially no force holding those surfaces together.

If the assembly floats in the right-hand position, its movement in that direction is stopped only when the roller 17 engages projections 19*a* and 20*a*, and roller 18 engages recesses 19*b* and 20*b*. However, such engagement of the cams and rollers has no effect on the engagement or disengagement of the clutch, nor does it apply any substantial frictional load to shaft 5.

The clutch members 6 and 7 are disengaged only when the diametrically opposite followers 17 and 18 move into engagement with diametrically opposite portions of the projections 19*a* and 20*a*. In the normal operating condition of the mechanism of Figs. 1 to 4, the cams 19 and 20 are positioned angularly relative to each other so that at least portions of the projections are diametrically opposite. When the followers 17 and 18 engage those diametrically opposite portions of the projection, the yoke 16, the pivot pin 15 and the shaft 5 are shifted bodily to

the left from the positions shown in Fig. 1 to the positions shown in Fig. 2. During this movement, the periphery of clutch plate 7 engages stop plate 28, thereby stopping the driven clutch plate 7 from following clutch plate 6 so that the clutch plates are separated. This engagement of the stop plate 28 by the periphery of clutch plate 7 is effective not only to separate the clutch plates 6 and 7, but also the friction between the stop plate 28 and the clutch plate 7 has a braking effect on that clutch plate, so that the output shaft 11 and its load are stopped from rotating substantially immediately as soon as the clutch is released by simultaneous engagement of the rollers 17 and 18 with the cam projections.

In this position of the parts, spring 13 is effective to hold the periphery of clutch plate 7 in engagement with the stop plate 28, and is also effective to hold the followers 17 and 18 in engagement with the cams 19 and 20.

On each of the cams 19 and 20, the projection extends through 180° of the periphery and the recess extends through the remaining 180°. With this arrangement, the cam 19 may be adjusted relative to the cam 20 from one extreme position shown in Fig. 3A, wherein the projections of the two cams are diametrically opposite each other, to an opposite extreme position shown in Fig. 3B, wherein the projections are aligned with each other. A half-way position is illustrated in Fig. 3C, wherein the projections of the two cams are offset by 90°.

When the two cams are in the relative positions shown in Fig. 3A, both rollers 17 and 18 are always engaging a cam projection, so that as the driving shaft 5 rotates, it is continuously maintained in the axial position shown in Fig. 2, wherein the clutch plates 6 and 7 are separated and the output shaft is not driven.

When the cam 19 is shifted to the position shown in Fig. 3B, one of the two rollers is always engaging a recess, so that the driving shaft 5 remains in the position shown in Fig. 1, the clutch plates 6 and 7 remain in engagement, and the output shaft 11 is continuously driven.

When the cam 19 is adjusted to the position shown in Fig. 3C, the rollers 17 and 18 are operated in a repeated time sequence such that the clutch is operated in a cycle including a period during which both rollers engage projections and the clutch is disengaged followed by a complementary period during which one roller engages a recess and the clutch is engaged. If the rollers 17 and 18 start in the positions shown in solid lines in Fig. 3C and move to the positions shown at 17*a* and 18*a* in dotted lines, it may be seen that the right-hand roller 18 will remain in a recess until it reaches the leading edge of the projection 19*a*. When it moves up on the projection 19*a*, both rollers will be on projections, since the left-hand roller 17 remains on the projection 20*a*. This condition continues for 90° of rotation of the shaft, until the left-hand roller 17 passes off the end of projection of 20*a*. The left-hand roller 17 remains in the recesses for another 90° of rotation of the shaft, at which time it, in turn, is lifted by the projection 19*a*. With this arrangement, it may be seen that the clutch plates 6 and 7 remain engaged for 90° of rotation of the drive shaft 5 and then are disengaged for 90° of rotation of that shaft. This cycle is repeated each half-revolution as long as the drive shaft rotates.

The proportional relationship between the complementary periods of clutch engagement and disengagement may be varied by adjustment of the angular position of cam 19. Fig. 4 shows the cams in a position where the cam 19 has been offset by an angle less than 90 from the position of the inner cam 20. It will be readily understood that by shifting the outer cam 19 with respect to the inner cam 20, the magnitudes of the angles in each half-revolution of the driving shaft through which both rollers 17 and 18 remain continuously engaged with projections may be varied from 0°, as shown in Fig. 3B, through 90°, as shown in Fig. 3C to 180°, as shown in Fig. 3A. Consequently, for a given fixed speed of the input shaft 2, the average speed of the output shaft 11 may be varied from zero to a maximum value determined by the ratios between the gears 3, 4 and 9, 10. For any multiple of half-revolutions of the driving shaft 5, the output shaft 11 may be accurately driven through a corresponding number of half-revolutions whose proportion with respect to the input shaft revolutions is determined by the setting of the hand wheel 25. The register 27 may be conveniently arranged to read the percentage of output revolutions to input revolutions (taking into account the gear ratio between the input and output shafts).

Although the cam adjusting mechanism is illustrated as being operated by a hand wheel 25, it will be readily recognised that any other suitable operating device may be substituted, including various types of automatic control.

The cam contours may be varied from those illustrated as determined by the requirements of any particular installation. For example, the projection of one cam may be made shorter than 180°, while the projection of the other cam is made equal to 180°. With such an arrangement, it would be impossible to adjust the cams so as to secure continuous rotation of the output shaft. Alternatively, the projection on one cam could be made longer than 180°, while the projection on the other cam is made equal to 180°. With such an arrangement,

the cams could not be adjusted so that the output shaft would remain stationary.

Another possible alternative would be to make one cam projection angularly greater than 180°, and the other cam shorter than the complementary angle required to complete the full circle. The output shaft would then have a minimum average speed greater than zero and a maximum average speed less than continuous rotation. In this way, any desired limits could be applied to the range of proportions of the input shaft speed at which the output shaft is operated.

The braking arrangement illustrated might be modified in several respects. The rotating braking surface need not be on the periphery of the clutch plate 7, as illustrated, but may be on any part which rotates and moves axially with the driven shaft 8. If desired, a non-braking stop arrangement could be used for limiting the axial movement of the driven shaft 8 instead of the braking stop arrangement shown. Such a non-braking stop would be used with a load which tends to brake itself.

The floating arrangement of the shafts 5 and 8, as described above, reduces the wear on the cams and rollers, since those parts are loaded only when both rollers 17 and 18 are engaging projections on the cams. If the floating action is undesirable for any reason, another biasing spring can be provided, for example, between the left-hand side of casing 1 and the gear 9, so as to hold the rollers 17 and 18 continuously in engagement with the cams.

Fig. 5 illustrates a modified form of variable driving apparatus in which the clutch is engaged when both rollers 17 and 18 are riding on cam projections, rather than being disengaged at that time as in the construction shown in Figs. 1 to 4. In this modification, the driving shaft is axially biased continuously so that the rollers 17 and 18 always ride on the surfaces of the cams 19 and 20.

In Fig. 5, those parts which correspond in structure and in function to their counterparts in Figs. 1 to 4 have been given the same reference numerals as those counterparts and will not be further described.

In Fig. 5, the gear 3 drives a gear 40 fixed on a driving shaft 30 which is journalled in the right-hand side of the casing 1. The driving shaft 30 corresponds generally to the driving shaft 5 of Fig. 1, and supports the yoke 14 by means of the pivot pin 15 in a similar manner.

Outside the casing 1, the end of shaft 30 rotatably supports a spring retainer 31 by means of a bearing 32. A compression spring 33 is held between the retainer 31 and the right-hand side of casing 1. The spring 33 biases the shaft 30 to the right, so that the rollers 17 and 18 are continuously held in engagement with the cams 19 and 20.

The driven shaft 34 is aligned with the driving shaft 30 and is journalled in the left-hand side of the casing 1. On its right-hand end, the shaft 34 carries a gear 35 which meshes with the gear 10 on the output shaft 11. The right-hand face of gear 35 serves as one plate of a clutch, the other face being formed by a clutch facing ring 36 attached to the left-hand side of gear 40. Near its hub, the gear 35 carries on its left-hand side a bearing 37. A spring 38 is retained in compression between bearing 37 and the left-hand wall of casing 1. The left-hand end of shaft 34 carries a stop flange 39. As shaft 34 is moved to the right by spring 38, flange 39 engages the left-hand wall of casing 1 and limits that movement.

The operation of the mechanism shown in Fig. 5 corresponds generally to that shown in Figs. 1 to 4, with certain exceptions including those noted above, namely, that the clutch is engaged when the rollers 17 and 18 are both engaging cam projections, and that the rollers 17 and 18 are continuously biased into engagement with the cams 19 and 20. Another difference in the operation is that there is substantially no braking action between the flange 39 and the left-hand side of the casing 1.

Thus when both the rollers 17 and 18 are riding on cam projections, the shaft 30 is moved to the left against the action of the spring 33 so as to engage the clutch and also to move the driven shaft 34 slightly to the left against the action of the spring 38 and thus to free the stop flange 39 from the left-hand wall of the casing 1.

The mechanism shown in Fig. 5 has been successfully used to drive a counter-weight along the lead screw of a scale beam. The apparatus shown in Figs. 1 to 4 has been used to drive a dry feeder of the type illustrated in the co-pending Application No. 11.731/53 (Serial No. 724,475) previously mentioned. It is, therefore, apparent that the present invention is capable of application to a wide range of loads, and that it is not limited to any particular field of use.

What we claim is:—

1. Variable speed driving mechanism including a clutch interposed between a driving shaft and a coaxial driven shaft and engaged and disengaged by relative axial movements of the two shafts, in which one of the shafts is reciprocated so as to engage the clutch for a proportion only of each revolution by means of a cam and follower device adjustable so as to vary the proportion of a revolution during which the clutch is in engagement.

2. Mechanism according to Claim 1, in which the adjustment may be carried out continuously over a range lying between 0

and 100 per cent of a revolution.

3. Mechanism according to Claim 1 or Claim 2, in which the driven shaft is braked during each interval when the clutch is disengaged.

4. Mechanism according to Claim 3, in which the braking of the driven shaft and the disengagement of the clutch occur simultaneously.

5. Mechanism according to Claim 3 or Claim 4, in which the two shafts are so biased as to press the clutch members together, and the driving shaft is reciprocated so as to engage and disengage the clutch, a stop member being provided which prevents the driven shaft following the driving shaft into one of its axial positions, with the result that the clutch members are separated against the effect of the bias and at the same time the driven shaft is braked.

6. Mechanism according to Claim 5, in which the two shafts are biased by means of a compression spring situated between flanges on the shafts.

7. Mechanism according to any one of Claims 1 to 4, in which the driving shaft is spring biased in a direction such as to separate the clutch members and the driven shaft is spring biased into contact with a stop in a direction such as to engage the clutch member, with the result that when the driving shaft is moved against its spring to engage the clutch, the driven shaft is moved away from its stop against its corresponding spring.

8. Mechanism according to any one of the preceding claims, in which the follower device comprises a yoke pivoted to the driving shaft to turn about an axis perpendicular to that of the shaft and carrying a pair of diametrically opposite cam followers co-operating with a pair of circular end-cam surfaces concentric with the driving shaft and lying one outside the other, the cam surfaces each being narrower than the radial width of the cam followers which thus overlap both surfaces; each cam surface is formed with a projecting portion and a recessed portion, the cams being rotatably adjustable in relation to one another so that when a recessed portion of one surface lies adjacent a projecting portion of the other surface the latter determines the position of a cam-follower bearing against it and the arrangement is such that when, during a revolution of the driving shaft, both cam followers are bearing against a projecting portion the driving shaft takes up an axial position in which a spring is compressed, while in all other positions of the cam followers the spring moves the driving shaft to another axial position so as to produce the required reciprocation.

9. Mechanism according to Claim 8, in which the projecting and recessed portions on each of the cam surfaces each extend over a semi-circle and one of the cams is adjustable angularly through one hundred and eighty degrees, so that when the projecting portions are diametrically opposite one another the shaft is maintained continuously in one axial position while when they are alongside one another the shaft is maintained continuously in the other axial position.

10. Mechanism according to Claim 1, constructed and operating substantially as described with reference to Figs. 1, 2, 3 and 4, or with reference to Fig. 5 of the accompanying drawings.

For the Applicants.
GILL, JENNINGS & EVERY,
Chartered Patent Agents,
51/52, Chancery Lane, London, W.C.2.

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Fig. 1

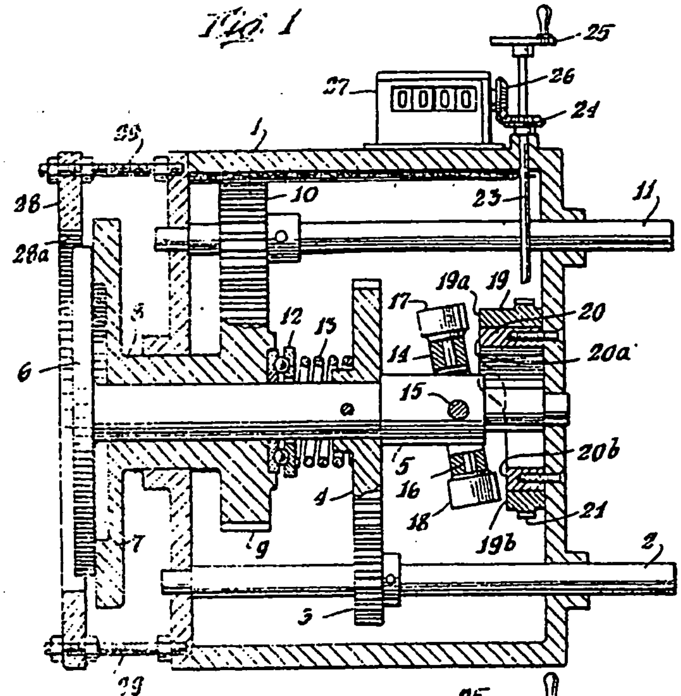


Fig. 2

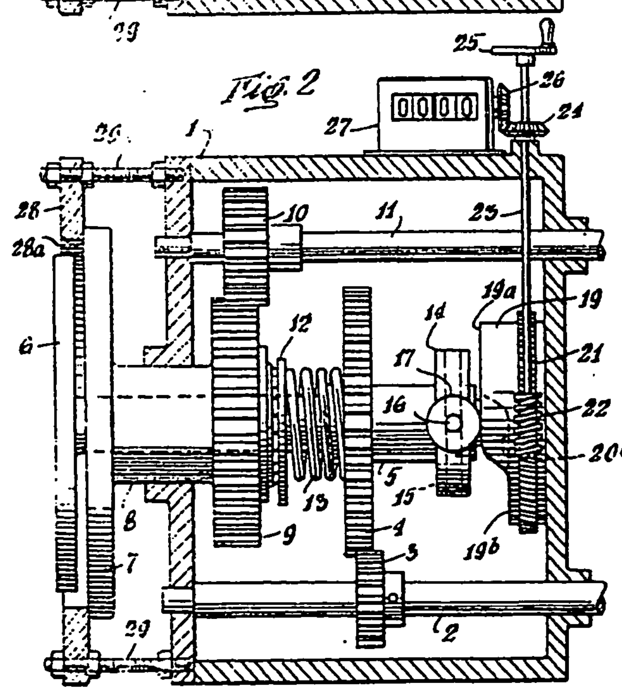
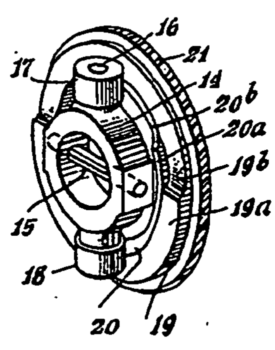


Fig. 4



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2 SHEETS

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SHEETS 1 & 2

Fig. 3

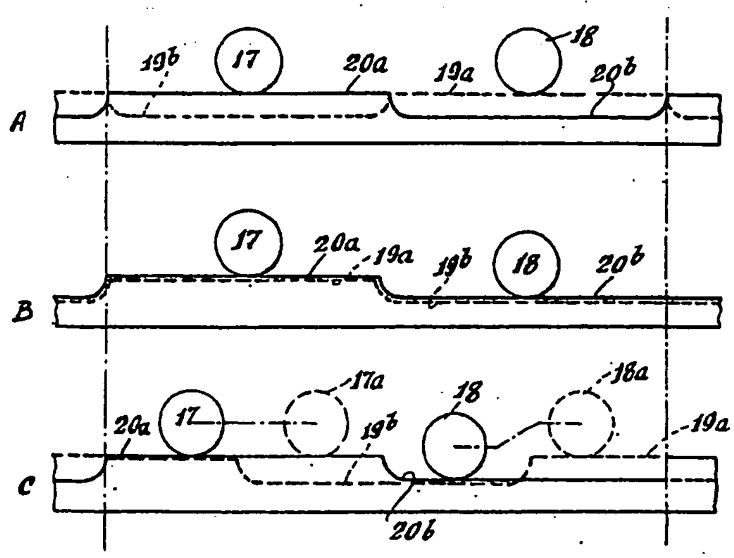


Fig. 4

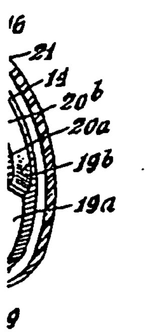
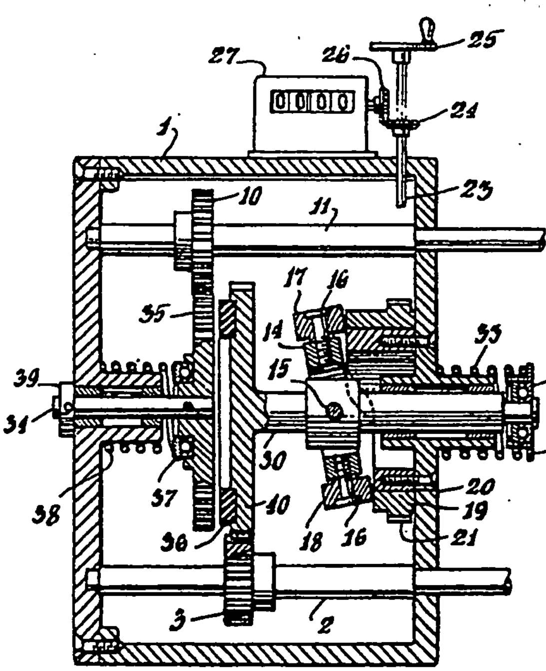


Fig. 5



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2 SHEETS
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SHEETS 1 & 2

Fig. 1

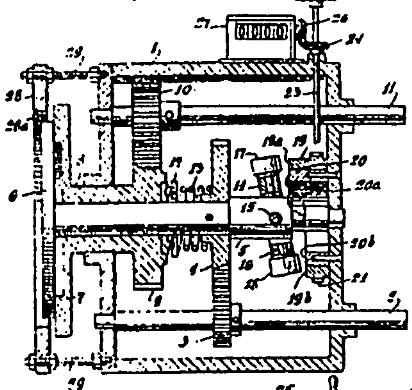

Fig. 2

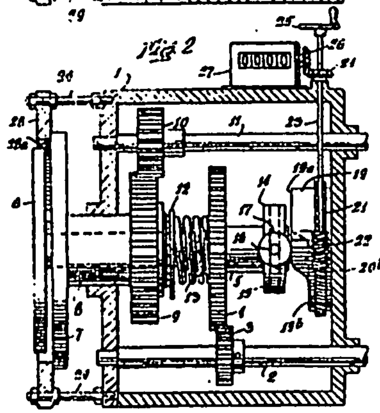

Fig. 4

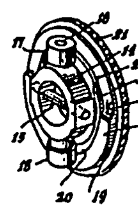

Fig. 3

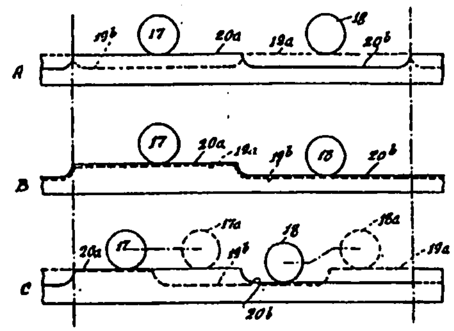

Fig. 5